# Developments in Control of Time-Delay Systems for Automotive Powertrain Applications

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***Abstract :-*** To provide an overview of several application problems in the area of automotive powertrain control, which can greatly benefit from applications of analysis and control design methods developed for time-delay systems. The application considered concern, the idle speed control (ISC) in gasoline engines. The nature of the delay and the role played by them in these application is highlighted and the imposed performance limitations are discussed. Links are provided with theoretical literature on the analysis and control of time-delay systems and modeling details are discussed to a level that will permit other researchers to use the associated models for simulation case studies.

***Keywords:-*** *Air-to-fuel ratio control, Automotive powertrain control, Linear matrix inequality, Lyapunov function, Observer design, Idle speed control.*

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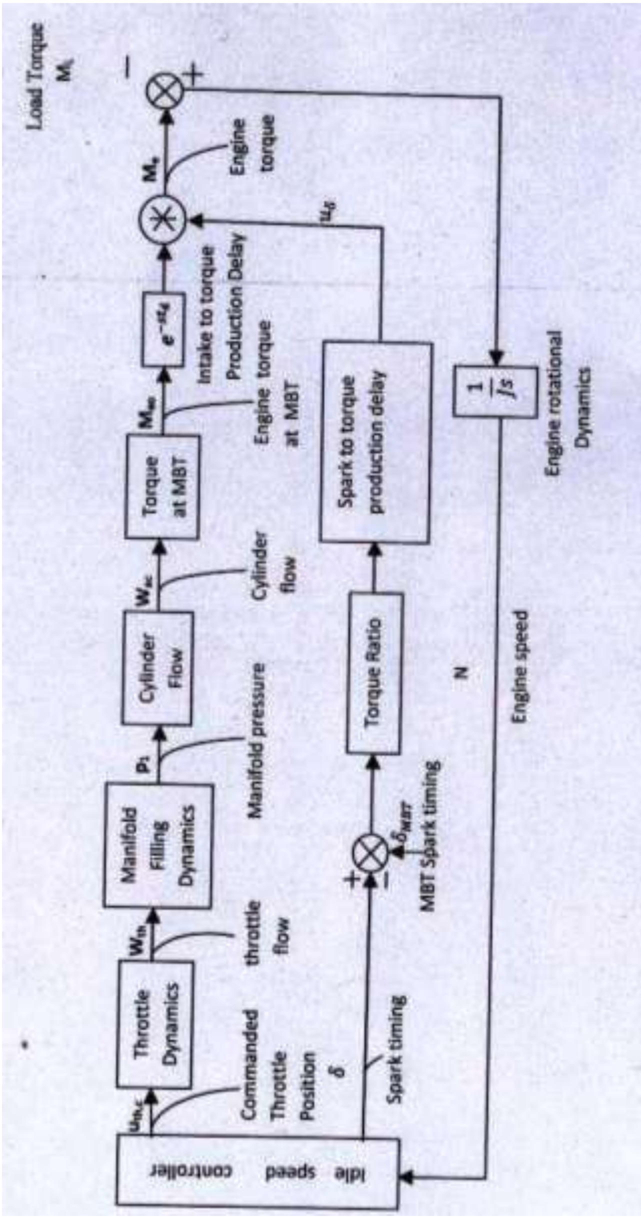
## INTRODUCTION

This application focus on problems in the area of automotive powertrain control, where the issues related to the treatment of time-delays constitute important design considerations. New engines and transmissions are introduced every year into production, and a broad range of advanced technologies are being continuously developed for future applications. Powertrain control is playing an increasingly important role in the automotive development cycle, and is a key enabler for future improvements in fuel economy and emissions. Since delays are common in powertrain systems, effective techniques for controlling time-delay systems can have a substantial impact on these applications.

Here, powertrain control application with delays are discussed. The application concern, the idle speed control (ISC) and air-to-fuel ratio control in gasoline engines. We will also provide links with theoretical literature on analysis and control of time-delay systems and discuss modeling details to a level that will permit others to use the associated models for the simulation case studies.

In the ISC case, the delay arises because of finite time between intake and power strokes of the engine and is generally assumed to be half of the engine cycle. The fueling rate is determined according to the predicted airflow at the time of the intake stroke and this prediction can be made so accurate that the torque can be assumed to be proportional to the airflow delayed only by the intake to power delay but unaffected by the delays in the fueling system. In the air-to-fuel ratio control problem, the delays in the fueling system must be considered explicitly as this loop compensates for the air-to-fuel ratio disturbances. Finally, in the diesel engine, the delay is due to finite time between intake and exhaust strokes of the engine and can be assumed to be about three-fourth of the engine cycle.

ISC is one of the key control functionalities of modern gasoline and diesel engines. It maintains engine running when the driver foot is off the gas pedal. A schematic block diagram of the ISC closed-loop system for a gasoline engine is presented in Fig.




the position of the electronic throttle and the spark timing are control inputs and the engine speed is a controlled output. The engine speed is assumed to be measured (in reality, it is fairly accurately estimated from the crankshaft position sensor signal). The objective of ISC is to maintain engine speed at a given set-point (around 625 rpm in drive gear and 650 rpm in neutral gear during fully warm operation) despite measured and unmeasured torque disturbances due to power steering, air-conditioning turn on and off, transmission engagement, alternator load changes, and the like. Both throttle and spark have an effect on the engine torque and can compensate the effects of disturbances. Tight performance requirements are imposed to contain the engine speed dip (maximum engine speed decrease when disturbance hits) within 80 rpm and engine speed flare (maximum engine speed increase when disturbance hits) within 120 rpm.

Increasing engine throttle opening causes air flow through the throttle to increase. The increase in the throttle flow causes the intake manifold pressure to increase, which in turn increases the airflow into the engine cylinders. Larger engine torque is produced as the cylinder air flow increase is proportionally matched by the increase in the engine fueling rate. Adjusting the spark timing shifts the start of combustion relative to the revolution of the crankshaft.

The spark timing (in degree of crankshaft revolution) at which the maximum torque is produced is referred to as maximum brake torque (MBT) spark timing. Typically, the spark timing can only be retarded relative to the MBT spark timing, which causes engine torque to decrease. The maximum feasible spark retard is limited by combustion stability constraints. To ensure that spark has a bidirectional authority over the engine torque, the set-point for spark-timing in steady-state is retarded from MBT. This steady-state spark retard is referred to as *spark reserve*.

The need to maintain spark reserve results in fuel consumption increase as compared with the case when spark timing is maintained at MBT. As we will see shortly, the delay in this system is a key factor that prevents making throttle to engine speed control loop faster and eliminating the spark reserve. Another opportunity to reduce fuel consumption during idling is to lower idle speed set-point, provided this can be sustained by vehicle accessories. Lowering idle speed set-point makes the problem harder in two ways. 1) the delay, which, as we will see shortly, is inversely proportional to engine speed, becomes larger. 2) the engine speed excursions, especially dips, must be much tighter controlled as engine speed decrease below the so-called “fishhook” point results in a rapid increase in engine friction (due to changing properties of the lubricant) and can easily produce an engine stall. Engine stall has a negative customer impact and the number of engine stalls is one of the key quality indicators (TGW–things gone wrong).

Historically, ISC is related to one of the oldest closed-loop systems discussed in the controls literature, the so called Watt’s governor (1787), which may be viewed as a speed controller for a steam engine. In older gasoline engines with a mechanical throttle (i.e., throttle mechanically connected to driver gas pedal), a dedicated actuator called air-by-pass valve was used for ISC but in modern engines the introduction of an electronic throttle, decoupled from the driver’s gas pedal, has permitted to eliminate the air-by-pass valve. This did not make the control problem easier as electronic throttle designed for regular driving operates only in a narrow range during idle where small imperfections may have large impact. Robustness to changes due to aging and manufacturing tolerances, operation across a wide range of environmental conditions (ambient temperature, pressure, humidity, etc.) and constant pressures to lower the cost of the hardware components (and achieve same quality of control with less capable components) are some of other sources of continuing challenges for ISC.

**A Model for ISC -**

A simplified model, based on the work of discussed to illustrate the delay-related aspects of ISC. The model is a continuous-time, mean-value model in which cyclic fluctuations of engine operating variables are replaced by their cycle averages. In addition, several approximations are made that are valid locally around idle operating point. With these approximations, the engine speed dynamics have the form $\dot{N} = \frac{1}{J/(30/\pi)}(M_e - M_L)$ ……..[1]

where $N$ is the engine speed in rpm, $J$ is the engine inertia in kg $m^2$, $(30/\pi)$ is a conversion factor from rad/s to rpm, $M_e$ is the torque produced by the engine (Nm), and $M_L$ is the load torque in Nm. The engine torque is represented as a product of engine torque at MBT, $M_{eo}$, and torque sensitivity to spark (also referred to as the torque ratio), $u_\delta$, which is a function of spark retard $M_e(t) = M_{eo}(t - t_d)u_\delta(t)$……..*[2] w*here $t_d$ is the delay between the intake stroke of the engine and torque production.Hereafter, we assume that we can manipulate $u_\delta$ as if it was a control input and we neglect the delay between spark timing and torque production, noting that omitting the spark to torque delay, while reasonable, is not uniformly agreed.The engine torque at MBT spark timing, $M_{eo}$, is a function of cylinder flow, $W_{ac}$ (kg/s), and engine speed, $N$: $M_{eo} = \frac{k_1}{N} W_{ac}$………..[3] where $k_1$ is a parameter. The cylinder flow is a function of the intake manifold pressure, $p_1$ (kPa) and engine speed, $N$, $W_{ac} = \frac{k_2}{k_1} p_1 N + k_0$………………[4] and $k_0$, $k_2$ are constant parameters.

The intake manifold filling dynamics are based on the ideal gas law and constant gas temperature(isothermal) assumption: $p_1 = \frac{RT_1}{V_1}(W_{th} - W_{ac})$…….[5] $R$ is the where ideal gas constant in kJ/kg/K, $T_1$ is the temperature of the air in the intake manifold in K, $V_1$ is the intake manifold volume, and $W_{th}$ is the throttle flow in kg/s. Near idle, the flow through the throttle is choked and one can approximate $W_{th} = k_3 u_{th}$, ……[6] where $u_{th}$ is the throttle position in degree and $k_3$ is a constant. From [3] and [4] it follows that $M_{eo} = k_2 p_1 + \frac{k_1}{N} k_0$ Differentiating this expression, we have, $\dot{M}_{eo} = -k_2 \frac{RT_1 N}{V_1 K_1} M_{eo} + k_2 \frac{RT_1}{V_1} K_3 u_{th} - \frac{k_0 k_1}{N^2} \dot{N}$ The term $\frac{k_0 k_1}{N^2} \dot{N}$ ……[7]

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is small and can be omitted so that

$\dot{M}_{eo} = -k_2 \frac{RT_1 N}{V_1 K_1} M_{eo} + k_2 \frac{RT_1}{V_1} K_3 u_{th}$ ……..[8] From [1] and [2] it now follows that

$\dot{N} = \frac{1}{J/(30/\pi)} (M_{eo}(t - t_d) u_\delta - M_L)$ …….[9] The complete model is now defined by [8] and [9] .

The delay $t_d$ is between the intake stroke of the engine and torque production, and is about $360^o$ of crankshaft revolution. Consequently, $t_d$ is inversely proportional to engine speed and is given by $t_d(t) = \frac{60}{N(t)}$ ……..[10]

Note that the delay is state-dependent and time-varying since $N(t)$ varies with time. In the practical treatment of the problem a constant delay assumption, with the delay value corresponding to that at the idle speed set-point, is often made. Some references cite a range between $230^o$ and $360^o$ for the delay duration in the crank angle domain rather than a fixed value of $360^o$, others augment to $t_d$ also the throttle delay and computing delay in the engine control module. For instance, in [14] the value of the delay is estimated as $450^o$ for a prototype engine, due to computations scheduling in the engine control module.

It is also interesting that transforming the model to the crank angle domain by defining $\theta = \frac{N(t)}{60} t$ ………[11] and using the relation

$\frac{d(.)}{d\theta} = \frac{60}{N} \frac{d(.)}{dt}$ renders [8] - [9] as $\frac{dM_{eo}}{d\theta} = \frac{60}{N}\left(-k_2 \frac{RT_1}{V_1} \frac{N}{k_1} M_{eo} + k_2 \frac{RT_1}{V_1} k_3 u_{th}\right)$ ……………[12]

$\frac{dN}{d\theta} = \frac{60}{JN/(30/\pi)} (M_{eo}(\theta - 1) u_\delta - M_1)$ …[13]

This is a plant model with ant delay equal to 1.

For a Ford F-150 vehicle, the model parameters in the above model have been estimated as follows [28, 47]: $k_1 = 3,961$,

$k_2 = 102$, $k_3 = 2.02$, $\frac{RT_1}{V_1} = 0.0750$ and

$J/(30/\pi) = 0.027$ while the nominal throttle position, load torque and engine speed are, respectively, $u_{th,0} = 3.15$, $M_L = 31.15$ and

$N = 800$. It is of interest to examine the linear zed transfer function at idle operating conditions from throttle position in degree to engine speed in rpm. It can be shown that this transfer function has the following form,

$$H(s) = \frac{256.867 X 2.228}{s^2 + 1.545 s + 2.228 e^{-t_d s}} e^{-t_d s} .$$

The step response of this transfer function has a lightly damped character (damping ratio about 0.5). The liberalized transfer function at idle operating conditions from load torque in Nm to engine speed in rpm has the form

$$H_L(s) = \frac{-37.04s - 57.22}{s^2 + 1.54s + 2.228e^{-t_d s}}$$

A first-order transfer function with a unit static gain may additionally be augmented to the above model to represent the dynamics of the actual throttle position, $u_{th}$, in response to the commanded throttle position, $u_{th,c}$ set by the idle speed controller. Reasonable parameter estimates for this throttle dynamics model are a static gain of 1, the time constant of about 0.05s and a delay of about 0.05s (if not already accounted in $t_d$).

## Concluding Remarks -

In this paper, I have discussed application problem in the area of power train control from the perspective of control of time-delay systems. The associated models and control problems for ISC and air-to-fuel ratio control in gasoline engines and for state estimation in diesel engines with exhaust gas recirculation and variable geometry turbo charging have been reviewed. Modern engines and transmissions rely on many control functions that are impacted by the delays and in this light we have only covered a small (but representative) subset of relevant application problems.

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